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Valve control for varying the valve stroke and duration of valve opening

A two-armed actuator lever (S) is in continual contact with the bearing surfaces of two cams (N_1, N_2), which are different in terms of their stroke and overall duration of stroke, and transfers the stroke movements of the cams to the valve via a transmitter ram (A_{st}). Spring-loaded tappets (F) intensify the force of the actuator lever (S) acting on the cams (N_1, N_2). A rotation of the control shaft (Sw), whose crank pin (Kz) slides in the guide slot (Fu) of the actuator lever (S), displaces the axis of rotation (D) of the actuator lever (S) and thus changes the lever arm ratio between the points of action of the cam strokes (H_1, H_2) of the two cams (N_1, N_2) and the axis of rotation (D). The proportion of the stroke of each of the cams, which contributes to the stroke of the valve varies, by which means a continuous variation of the maximum valve stroke and the duration of the opening of the valve is obtained. Thus the control times and the control time cross-sections can be adapted to the requirements of different engine speeds and loads of internal combustion engines. In the foreground of this invention is the ability to optimize the variation in torque performance over the engine speed range in order to increase the economy and environmental compatibility of an engine.

Patent Claims

1. Valve control for varying the valve stroke and duration of valve opening, **characterized in that**:

a) the lift curve of a valve is produced by two cams (N_1, N_2), which differ in their maximum stroke and their overall duration of stroke and which both rotate at the same speed in the same or opposite sense about their, in particular, mutually parallel axes of rotation.

b) the bearing surface of each cam is in continual contact with one limb of a two-armed lever, referred to in the following as the actuator lever (S), whose axis of rotation or point of rotation (D) transfers the stroke movement, which results from the stroke movements of the two cams, to the valve, either directly, or via a transmitter ram, e.g. a tappet (A_{st}), a drag lever (A_{sh}) or a valve lifter (A_k).

c) by varying the lever arm ratio (H) of the respective distances between the points of action of the strokes of the two cams on the actuator lever (H_1, H_2) and the axis or point of rotation (D) of the lever, the situation is achieved in which the proportion of the stroke of each cam to the stroke on the valve can be varied.

2. Valve control according to Claim 1, characterized in that, in addition to the valve spring, the continual contact of each of the two cams with the actuator lever (S) can be maintained by springs or spring-loaded additional elements (F), e.g. tappet or plunger elements in contact with the actuator lever.

3. Valve control according to Claim 1, characterized in that in order to vary the lever arm ratio (H), either the actuator lever (S) or the bearing arrangement of the transmitter ram (A_{sh}, A_{st}, A_k) is displaced, or an additional intermediate link (Z), e.g. a sliding block, is arranged between the actuator lever and the transmitter ram and is displaced.

4. Valve control according to Claim 3, characterized in that:

a) the displacement of the actuator lever (S) can be provided by means of a control shaft (S_w).

b) in particular, this control shaft (S_w) is arranged parallel with the axes of rotation of the two cams (N_1, N_2) and has a crank pin (Kz), which slides

in the groove or slot-type guide (Fü) of the actuator lever (S) and on rotation of the control shaft exactly positions the axis of rotation (D) of the actuator lever.

c) the selected position of the axis of rotation (D) of the actuator lever also remains at least approximately in the same position during the stroke movement of the actuator lever, i.e. in that the guide slot or groove (Fü) is, in particular, arranged at right angles to the face of the actuator lever that remains in contact with the cams (N_1, N_2).

5. Valve control according to Claim 1, characterized in that the form and arrangement of the two cams (N_1, N_2), of the actuator lever (S), of the transmitter ram (A_{sh}, A_{st}, A_k), of all other components contributing directly to the variation of the lever arm ratio (H) and, if provided, the intermediate link (Z) and the springs or spring-loaded additional elements (F), are determined in such a way, that incorrect opening of the valve and an impermissibly significant increase in the valve play do not occur between the actuator lever (S) and the transmitter ram (A_{sh}, A_{st}, A_k) due to a variation in the lever arm ration (H) during the phase, in which the valve should be completely closed.

6. Valve control according to Claim 1, characterized in that insofar as the two cams (N_1, N_2) have axes of rotation that run parallel with each other, these can be arranged to be offset in the axial direction, such that the previously smallest possible distance between the axes of rotation can be even further reduced.

7. Valve control according to Claim 1, characterized in that the force transfer surface between the actuator lever (S) and the transmitter ram (A_{sh}, A_{st}, A_k) can be provided with a form similar to that of the cam, so that the inclined attitude (α) of the actuator lever (S), resulting from the different magnitudes of the stroke movements of the two cams, can be utilized for further targeted changes in the valve stroke at specific sections of the valve lift curve.

8. Valve control according to Claim 1, characterized in that by using a rocker lever (A_k) or drag lever (A_{sh}) as the transmitter ram, with a variation of the lever arm ratio (H) on the actuator lever (S), the lever arm ratio on the rocker lever (A_k) or on the drag lever (A_{sh}) can be varied and that by this means an additional variation of a particular already known variation of the valve lift curve is obtained, which is superimposed on the variation according to Claim 1c) to

reinforce it advantageously, in that the axis of rotation of the cam with the largest maximum stroke (N_2) lies closer to the axis of rotation of the rocker lever (A_k) or drag lever (A_{sh}) than the axis of rotation of the other cam (N_1).

Description

The invention described in the following relates to the technical field "Variable valve controls in internal combustion engines".
A number of designs have already been published on this topic.
These designs can be sub-divided into the following groups, in which this invention belongs in the first group:

a) Systems, which vary the valve stroke and the duration of valve opening

- bevelled cams, cams or secondary matching components axially displaceable on the cam axis of rotation (see List of References, Nos. 1 – 4)
- oil-filled chamber (hydraulic tappet) between cam and valve, variation of the oil volume through controlled bores (see List of References, Nos. 5 – 9)
- rotary or translatory [or reciprocatingly] oscillating cams with adjustable intermediate elements between cam and valve (see List of References, Nos. 10, 25, 4)
- stroke camshaft plus control camshaft, which enables a displacement of the rocker lever axis of rotation and thus a premature regulation of the valve lift (see List of References, No. 11).

b) Systems, which only vary the duration of valve opening

- one cam for respectively producing the opening and closing curve of a valve, both cams being able to rotate relative to each other (see List of References, No. 12, 13)
- unequal cam angular velocity (see List of References, Nos. 14 – 17)
- magnetic valve actuation (see List of References, No. 18)

Systems, which only vary the valve stroke

- variation of the rocker lever or drag lever transmission ratio (only applies to the theoretical case, in which there is no valve play or elasticity in the valve drive, or otherwise a variation in valve opening duration also occurs) (see List of References, No. 19)

c) Systems, which directly vary neither the maximum valve stroke nor the duration of valve opening

- Rotational displacement of the cams relative to the crankshaft, i.e. displacement of the valve lift curve parallel to the time axis (see List of References, Nos. 20 – 23)
- Arrangement of additional control devices (sliders, flaps, or the like) ahead of the inlet valves or behind the outlet valves, whose variably controlled opening cross-sections are superimposed on the constant opening cross-sections of the conventionally controlled inlet and outlet valves (see List of References, No. 24).

The valve control times and the time cross-sections included from the valve lift curve are decisive in determining the characteristic of an internal combustion engine in respect of performance, economy and exhaust gas composition.

It is customary in engine construction to use camshafts with fixed control times, by means of which a constant valve lift is produced in valve stroke and opening duration (expressed in ° Kw [= degrees crankshaft]) over the entire operating range of the engine.

However, such a method of valve control cannot take optimum account of the requirements of various engine load and speed ranges in respect of different control times. In fact a compromise is always made in favour of the principal area of application of the engine.

The basic idea of the invention is therefore to be able to infinitely variably select during the operation of the engine between two strongly contrasting forms of cam in order to combine different engine characteristics in one engine.

The main task of the invention is the optimization of the torque and power curves of an internal combustion engine, in particular of a spark ignition engine at full load, as a function of engine speed:

- on engines with very high specific output [i. e. high output per cubic displacement] a satisfactory full load torque can be achieved by a reduction in the valve stroke and the control time cross-sections, even in the lower rpm range.
- on engines of medium specific output the torque can be increased at full load, in both the low and high rpm ranges.

The increase in torque generally results in a reduction in specific fuel consumption, since the engine can output the same power at a lower rpm and therefore at a higher mechanical efficiency.

Assuming application in a series production vehicle engine, for which the invention is principally conceived, it is particularly the increase in torque in the lower engine speed range that is of interest in terms of the commercial use of the invention:
By means of a longer gear transmission ratio the engine speed level can thus be reduced, whilst at the same time fuel consumption is reduced, noise level is reduced and also engine wear, without compromising the acceleration performance of the vehicle.

In addition to increasing torque, the invention can also be employed to improve the idling behaviour of spark ignition engines having a relatively long valve overlap phase. A reduction of the valve overlap time cross-section during idling and at low load reduces the residual gas (exhaust gas) fraction in the cylinder charge.

This results in an improvement in the thermal efficiency of the combustion process and stable idling performance at constant engine speed.

In addition, on the spark ignition engine it is possible to employ the invention in a targeted way in order to reduce specific exhaust gas pollutants: An increase in the valve overlap time cross-section, in particular at low engine load, leads to an “internal recirculation of exhaust gases”, which increases the

proportion of exhaust gas in the cylinder charge and reduces the peak temperatures of the combustion process.
By this means the content of environment-damaging nitrogen oxides (NO_x) in the exhaust gas is significantly reduced.

According to the invention these tasks are solved by a design with the characterizing features of Claim 1.

The main advantage of the invention lies in the almost exclusively mechanical construction of the components, which function uncritically both alone and in interplay with their paired components and, for the most part, have been proven over a long period of time.

By this means the possible technical problems associated with other systems for the variation of valve stroke and opening duration can be largely circumvented:

- very small contact surface between the cams and their matching components, which is hardly avoidable on bevelled cams, resulting in very high surface pressures and poor lubrication conditions, is not a feature of the invention.
- on hydraulic systems, pressure fluctuations in the oil chamber and problems with the fast feed and return of the oil at high engine speeds are possible.
- with oscillating cams, unfavourable lubrication conditions between the cams and matching components exist in the context of the reversal of movement of the cams (cam velocity equal to zero).

The invention enables a variation of the valve lift curve to be achieved, both in the region of the valve opening time points and in the region of the valve closing time points, and by using an appropriate design of cam this variation can be provided in only one of these regions. However, some systems (see, for example, Nos. 5, 9 and 11 in the List of References) are in principle only able to provide a variation either of the opening control time or of the closing control time.
An embodiment of the invention is presented in Figures 7 to 10 and is described in detail in the following:

Fig. 7 shows a cross-section of the embodiment. The two different cams (N_1, N_2) are disposed on parallel shafts and are in contact by means of the actuator lever (S), which provides the cams with a continuous flat bearing surface. The actuator lever (S) has a curved raised area on its lower surface which forms the force transfer surface between the actuator lever (S) and the transmitter ram (A_{st}).

The force transfer surface in this case is a section of a cylindrical shell, in which the axis of the cylinder lies on the continuous flat bearing surface of the top of the actuator lever (S). By this means the arrangement is such, that when the actuator lever (S) is at an inclination α, in accordance with the invention, it has no additional influence on the valve stroke movement. (Such an additional influence could, however, be obtained by analogy with Claim 7, by an appropriate shaping of the force transfer surface.)

The force transfer surface on the underside of the actuator lever (S) thus acts simultaneously to the point of rotation D of the two-armed actuator lever and the (variable) lever arm ratio H is given in accordance with Claim 1c, as:

lever arm ratio H = (distance H_1 – D) / (distance H_2 – D)

(see also Fig. 1)

A bucket tappet with hydraulic valve play compensation is used to provide the load-transmitter ram (hydraulic tappet) (A_{st}), whose upwardly facing circular surface is in a plane parallel with the plane of the upper bearing surface of the actuator lever (S), with the valve closed. Claim 5 is satisfied by this means.

Since the rise and fall ramps (cam lead-in) of the larger cam (N_2) have an effect on the actuator lever S before and after those of the smaller cam (N_1) (see Fig. 11), cam (N_2) is responsible for overcoming the valve play during the lift and reseating of the valve.

On a shift of the axis of rotation D of the actuator lever S towards the smaller cam (N_1), the stroke required on the larger cam (N_2) to overcome the valve play increases by up to an order of magnitude of the valve play. It is for this reason that the limitation of the valve play by means of the hydraulic tappet

(A_{st}) is a practical solution, because otherwise the cam (N_2) would have to be designed with a very long and high lead-in, or the possible displacement path of the axis of rotation D would be restricted.

The hydraulic tappet (A_{st}) reproduces the stroke movement to the valve in accordance with Claim 1b). The hydraulic tappet and valve with spring, wedge pieces and valve spring retainers are of conventional construction; an increase in the area of the hydraulic tappet facing the actuator lever can be useful in increasing the possible displacement path of the actuator lever S.

Two spring-loaded auxiliary tappets F provide for the continual pressure of the actuator lever S on the cams (N_1, N_2), in accordance with Claim 2. Their spherical contact surfaces slide in corresponding grooves on the underside of the actuator lever S, to allow for its lateral movement.

The control shaft Sw displaces and positions the point of rotation D of the actuator lever S, as described in Claim 4. In this case, two crank pins Kz are provided per actuator lever S, which engage on two sides in respective guide slots Fü in the actuator lever S and are provided with rollers R to reduce the friction (see also Fig. 10).

The arrangement of the guide slot in accordance with Claim 4c) provides for a very small inherent displacement of the actuator lever S during its stroke movement, due to the then simultaneously occurring inclination α of the lever. This intrinsic displacement effect is difficult to avoid, however, it has only a minimal affect on the selected position of the point of rotation D.

The crank webs of the control shaft Sw can at the same time provide a means of lateral guidance of the actuator lever S, in that they surround its end with minimal play (Fig. 10 and Fig. 8).

The phase position of the two cams N_1 and N_2 relative to each other is not displaced in the example described, i.e. – if reference is made initially to Fig. 7- which shows that the points of the cams point vertically upwards and vertically downwards, at the same respective points in time (see also Fig. 11).

Fig. 8 shows the example of the embodiment in a plan view of the control arrangement for a multi-cylinder in-line engine with the valves arranged in-line (2 valves per cylinder). Both the inlet and outlet valves are provided with the variable control according to the invention. The through going camshafts are not

shown in the area of a cylinder, in order that the arrangement of the actuator lever S, the tappet A_{st} and the auxiliary tappet F can be identified. Corresponding to Claim 6, the cams N_1 and N_2 are offset in relation to each other. The minimal separation between the axes of the camshafts Nw_1 and Nw_2 is advantageous in a number of ways:

- The displacement path of the rotation axis D of the actuator lever S is limited by the area of the tappet surface. A minimal shaft separation is therefore favourable to obtain an adequately large range of variation of the lever arm ratio H on the actuator lever S.
- In the case of a coupling of the two camshafts Nw_1 and Nw_2 by means of a gear pair Zp the diameter of the gear wheels is reduced.
- The bending moments occurring on the actuator lever S are smaller, due to the reduced lever arms, so that the actuator lever can be made lighter.

The coupling of the two camshafts through the gear pair provides directions of rotation of the opposite sense, leading to the following advantages:

- The sliding friction forces, which arise between the cams N_1 and N_2 and the surface of the actuator lever, oppose each other and to a large extent cancel each other out. In this way the forces required to position and displace the actuator lever are reduced.
- The periodic migrations of the points of contact of the two cams (or lines of contact), B_1 and B_2 on the actuator lever surface (away from the perpendiculars to the actuator lever surface, due to the axes of rotation of the cams) are mostly opposed in alignment at the same point in time, i.e. either both outwards or both inwards towards the respective other cams. By this means the lever arms of the cam forces are more uniformly distributed from the axis of rotation D to the two points of contact B_1 and B_2, so that, in particular, the auxiliary tappets F can be equipped with weaker springs.

In addition, the pitching moments on the actuator lever S are thus reduced, since the line connecting the points of contact B_1 and B_2 between the cams and the actuator lever lie on average closer to the central axis of the hydraulic tappet.

On displacement of the axis of rotation (variation of the lever arm ratio H) to a position almost under the axes of rotation of the respective cams, the contact points B1 and B2 and hence the cam forces move across to act on one arm of the actuator lever S. In this case the springs in the auxiliary tappets F must be strengthened in order that Claim 2 remains fulfilled.

The continuous control shaft Sw, running parallel with the cam shafts, is supported in bearings behind each 2nd cylinder and extends out of the cylinder head at one end. The torque required for the adjustment of the actuator lever S and thus to provide the variation in the valve lift curves can be applied at this free end of the shaft by means of an electric motor drive, a hydraulic device or by mechanical means e.g. by centrifugal force actuation.

Fig. 9 shows the actuator lever S from the most important angles. The narrow webs on the upper side ensure additional lateral guidance of the actuator lever on the cam side surfaces and serve at the same time to increase the bending strength.

In the following, using the example of an inlet valve lift curve, the possible variation in the valve lift curve is presented on the basis of the embodiment (Figs. 7 – 10). To this end, two different smooth [or jerk-free] cams have been developed using the method of calculation of Dipl.-Ing. D. Kurz (List of References, No. 26), whose exact form and phase position are shown in Fig. 11.

The related valve lift curves can be calculated from the cam stroke curves and the given geometrical relationships for the two extreme settings of the actuator lever S at the respective end of the displacement path.

The possible maximum displacement path of the actuator lever S with the hydraulic tappet diameter of the embodiment (35 mm) is approximately 30 mm with a camshaft separation of 50 mm.

Elasticity in the force transfer elements has not been taken into account in the calculations.

Furthermore, it has been assumed that the hydraulic tappet A_{st} is compressed by 0.1 mm at the start of the valve opening, but otherwise acts as a quasi-rigid element, i.e. that there is a constant valve play of 0.1 mm between the actuator lever S and the hydraulic tappet A_{st} (see List of References, No. 27).

The embodiment accordingly enables a continuous variation of the inlet valve lift curve between the two curves presented in Fig. 12.

To provide a comparison, Fig. 13 shows the inlet valve lift curves of two quite different 'fixed' camshafts:

- The camshaft of a series production engine with somewhat above average specific output.
- A so-called "torque camshaft", which improves the torque at full load in the lower to medium rpm range, but however permits only a relatively low peak performance to be obtained from the engine.

The following control times result from the lift curves in Fig. 12:

	Inlet opens before TDC	Inlet closes after BDC	Overall control time
Curve 1	36° Ca	59° Ca	275° Ca
Curve 2	70° Ca	91° Ca	341° Ca

Note: 2° Ca (Crankshaft angle) = 1° Cam (rotation) angle.

The curve of the angle of inclination α, introduced in Claim 7, as a function of the cam angle of rotation φ, is presented in Fig. 14. This variation results from the form and phase position of the two cams of the embodiment (Fig. 11).

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Key to the figures

Figure 1:	Ventilschaftende = end of valve stem
Figure 11:	Nocken = cam Hub = stroke (or lift)
Figure 12:	Kurve = curve Hub = stroke (or lift)
Figure 13:	Serien-Welle = series production shaft, Drehmoment-Welle = torque-shaft
Figure 14:	Schräg-stellung = Angle of inclination α Grad = Degrees

Valve timing gear for varying valve lift and valve opening time

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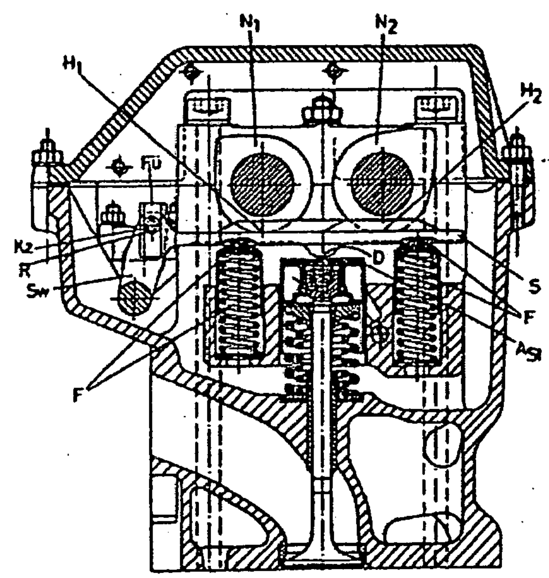
Abstract of **DE3725448**

A two-armed adjusting lever (S) is in constant contact with the running surfaces of two cams (N1, N2), differing in their lift and overall lift duration and transmits the cam lifting movements by way of a pickup (ASt) to the valve. Spring-loaded tappets (F) increase the contact pressure of the adjusting lever (S) on the cams (N1, N2). A rotation of the control shaft (Sw), the crank pin (Kz) of which slides in the guide groove (Fü) of the adjusting lever (S), displaces the axis of rotation (D) of the adjusting lever (S) and thereby adjusts the lever arm ratio between the lifting action points (H1, H2) of the two cams (N1, N2) and the axis of rotation (D). The share of the lift of each individual cam in the valve lift varies, resulting in a stepless variation of the maximum valve lift and the valve opening time. This allows the valve timing and valve timing cross-sections to be adjusted to the requirements of different engine speeds and loads in internal combustion engines. The primary concern is to improve the torque/speed curve in order to increase the economy and ecological safety of an engine.



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